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Review

An overview of current and future sustainable gas turbine technologies

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Abstract

In this work an overview of current and future sustainable gas turbine technologies is presented. In particular, the various gas turbine technologies are described and compared. Emphasis has been given to the various advance cycles involving heat recovery from the gas turbine exhaust, such as, the gas to gas recuperation cycle, the combined cycle, the chemical recuperation cycle, the Cheng cycle, the humid air turbine cycle, etc. The thermodynamic characteristics of the various cycles are considered in order to establish their relative importance to future power generation markets. The combined cycle technology is now well established and offers superior to any of the competing gas turbine based systems, which are likely to be available in the medium term for large-scale power generation applications. In small-scale generation, less than 50 MWe, it is more cost effective to install a less complex power plant, due to the adverse effect of the economics of scale. Combined cycle plants in this power output range normally have higher specific investment costs and lower electrical efficiencies but also offer robust and reliable performance. Mixed air steam turbines (MAST) technologies are among the possible ways to improve the performance of gas turbine based power plants at feasible costs (e.g. peak load gas turbine plants).

Keywords: Power generation; Gas turbine; Combined cycle

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1. Introduction

Gas turbines in simple-cycle mode have long been used by utilities for limited peak power generation. In addition, industrial facilities use gas turbine units for on-site power generation, usually in combination with process heat production, such as, hot water and process steam. In recent years, the performance of industrial gas turbines has been improved due to considerable investments in research and development, in terms of fuel-to-electricity conversion efficiency, plant capacity, availability and reliability. The greater availability of fuel resources, such as natural gas, the significant reduction in capital costs and the introduction of advance cycles, have also been a success factor for the increased deployment of gas turbines for base load applications [30].

In regard to the simple-cycle gas turbine technology, the major driver to enhance the engine performance has been the increase in process conditions (temperature and pressure) through advancements in materials and cooling methods. On going development and near term introduction of advanced gas turbines will improve the efficiency of the simple-cycle operation more than 40%. The combination of the gas turbine cycle (Brayton cycle) with a medium or low temperature bottoming cycle (like the Rankine cycle), known as the conventional combined cycle, is the most effective way to increase the thermal efficiency of a gas turbine cycle. Heavy duty natural gas fired gas turbines in combination with heat recovery steam generators and steam turbines represent the state of the art of this approach [32]. Inexpensive and readily available media (like air and water), well developed technologies (gas turbine, heat recovery steam generator, steam turbine), short construction time and, in particular, the high overall efficiency, have led to wide acceptance of this scheme. Combined cycle plants are already achieving efficiencies well over 58%, with plant capacities in the range between 350 and 500 MWe [31].

However, the development pace decelerated as the most readily available technical advances were exploited. Furthermore, in small-scale power generation (less than 50 MWe), it is generally more cost effective to install a less complex power plant, due to the adverse effect of the economics of scale. Combined cycle plants in this power output range have usually higher specific investment costs and lower electrical efficiencies but, on the other hand, robust and reliable performance. Thermodynamic cycle developments, such as recuperation, inter-cooling or after-cooling and cycle integration, such as mixed air steam turbines (MAST) are among the possible ways to improve the performance of gas turbine based power plants at feasible costs.

MAST technologies (also, referred as wet gas turbines [37] or as mixed gas steam turbines [27]) can improve the performance of a simple-cycle gas turbine by the integration of the bottoming water/steam cycle into the gas turbine cycle in the form of water or steam injection. Such configuration has a higher electrical efficiency than the simple-cycle gas turbine and produces more electricity per unit fuel input. Well known schemes of this technology are the steam injection gas turbines [4] and the humid air turbines [33]. The waste heat of the gas turbine is recovered and is utilised to produce steam, which is afterwards injected into the combustion chamber, or is used to humidify the compressed combustion air. Hence, the mass flow of the expanding flue gas is increased and, thereby, the fuel-to-electricity efficiency and the electrical power output are raised. This approach is favourable because water, unlike air, requires significantly less

compressor work. Nevertheless, it should be mentioned that the expansion of steam inside the gas turbine to atmospheric pressure is less efficient than inside a steam turbine, where the steam leaves the turbine at much lower pressures and, thus, provides more power and higher efficiency. Furthermore, the design of modern heat recovery steam generators includes two or more pressure levels and reheaters, which allows advanced recovery of the flue gas thermal energy. Therefore, a MAST technology will always have a lower efficiency than in combined cycle operation [10].

In this work, an overview of the current and future sustainable MAST and non-MAST gas turbine technologies is carried out. In particular, the various MAST and non-MAST gas turbine technologies are described and compared. Emphasis has been given to the various advance cycles involving heat recovery from the gas turbine exhaust, such as, the gas to gas recuperation cycle, the combined cycle, the chemical recuperation cycle, the Cheng cycle, the humid air turbine cycle, etc. The thermodynamic characteristics of the various cycles are considered in order to establish their relative importance to future power generation markets.

In Section 2, the gas turbine technology is briefly described, the various factors affecting the gas turbine performance are discussed and the gas turbine manufacturers are tabulated with their products specifications. The various non-MAST technologies are presented in Section 3 and in Section 4, a review of the various MAST technologies is given together with the technical and economic characteristics of the MAST gas turbines found during the market survey. In Section 5, the MAST technologies are critically compared with the conventional combined cycle technology. The conclusions are summarised in Section 6.

2. The gas turbine technology

In this section, the basic of the simple-cycle gas turbine are outlined. Then, the various factors affecting the performance of gas turbines are discussed. Emphasis is given on steam and water injection for power augmentation, which is the primary characteristic of MAST technologies.

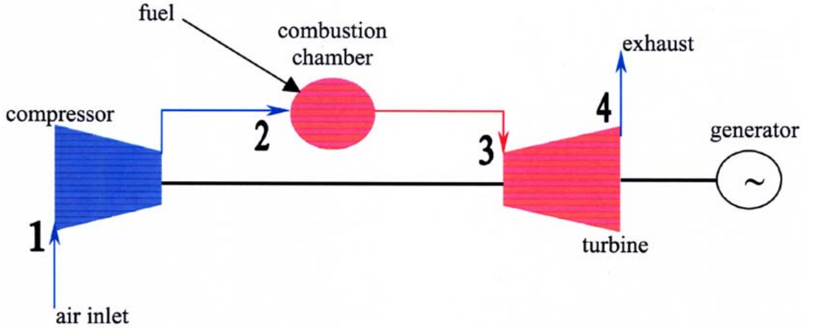

Fig. 1. The simple-cycle gas turbine.

2.1. The simple-cycle gas turbine

A schematic diagram for a simple-cycle gas turbine, for power generation, is shown in Fig. 1. Air enters the axial flow compressor at point 1 at ambient conditions. The standard conditions used, since these vary from day to day and from location to location, by the gas turbine industry are 15 °C, 1013 bar and 60% relative humidity, which are established by the International Standards Organization (ISO) and referred to as ISO conditions.

Air entering the compressor at point 1 is compressed to some higher pressure. No heat is added, however, compression raises the air temperature so that the air at the discharge of the compressor is at higher temperature and pressure, Upon leaving the compressor, air enters the combustion chamber at point 2, where fuel is injected and combustion occurs. The combustion process occurs at essentially constant pressure. Although high local temperature are reached within the primary combustion zone (approaching stoichiometric conditions), the combustion system is designed to provide mixing, burning, dilution and cooling. Thus, by the time the combustion mixture leaves the combustion system and enters the turbine at point 3, it is at a mixed average temperature. In the turbine section of the gas turbine, the energy of the hot gases is converted into work.

This conversion actually takes place in two steps. In the nozzle section of the turbine, the hot gases are expanded and a portion of the thermal energy is converted into kinetic energy. In the subsequent bucket section of the turbine, a portion of the kinetic energy is transferred to the rotating buckets and converted to work. Some of the work developed by the turbine is used to drive the compressor, and the remainder is available for useful work at the output flange of the gas turbine. Typically, more then 50% of the work developed by the turbine sections is used to power the axial flow compressor.

Although the exhaust is released at temperature of 400–600 °C and represents appreciable energy loss, modern gas turbines offer high efficiency (up to 42%) and a considerable unit power output (up to 270 MWe). Some typical modem gas turbines are listed in Table 1.

One important disadvantage is that a gas turbine does not perform well in part-load operation. For example, at 50% load, the gas turbine achieves around 75% of the full-load efficiency, and at 30% load this drops to 50% of the nominal efficiency. Therefore, arrangements, such as the controlled inlet guide vanes and multi-shaft designs, are employed to improve the part-load performance. Other modifications of the cycle include reheat, inter-cooling and recuperation. The expansion work can be increased by means of reheating. Moreover, this makes it possible to provide full-load efficiency within a broader load range by varying reheat fuel flow. Because of the increased specific work output due to reheat, the plant becomes compacter. Another technique to increase the specific work output is inter-cooling, which diminishes the work required by the compressor. The compressor outlet air becomes colder and, if air cooling is applied, this allows higher turbine inlet temperatures.

2.2. The Brayton cycle

The thermodynamic cycle upon which all gas turbines operate is called the Brayton cycle (or the Joule cycle). Fig. 2 shows the classical pressure versus volume (P–V) and

Table 1
Performance specifications of some modern gas turbines

Turbine	Manufacturer	Power (MWe)	Efficiency
SIA-02	KAWASAKI Heavy Industries	0.2	16.5
SIT-02	KAWASAKI Heavy Industries	0.4	16.2
PW6E	Ebara Corporation	0.6	20.9
PW6M	Ebara	0.6	22.5
PW7E	Ebara Corporation	0.7	22.0
S2A-01	KAWASAKI Heavy Industries	0.7	21.9
PW7M	Ebara	0.8	23.5
SB5	MITSUI Eng. & S.B. Co.	1.1	25.5
TGC105CS	Tuma Turbomach	1.1	24.2
Saturn 20	Solar	1.1	24.2
M1A-01	KAWASAKI Heavy Industries	1.2	20.9
PW12M	Ebara	1.3	22.5
M1A-11	KAWASAKI Heavy Industries	1.3	24.5
M1A-13CC	KAWASAKI Heavy Industries	1.4	22.3
KG2-3C	Dresser-Rand	1.5	15.8
M1A-03	KAWASAKI Heavy Industries	1.5	21.9
M1A-13	KAWASAKI Heavy Industries	1.6	25.5
PW14M	Ebara	1.6	23.5
Hurricane	Europcan Gas Turbines	1.6	24.5
KG2-3C	Dresser-Rand	1.8	16.0
KG2-3E	Dresser-Rand	1.9	16.5
PGT2	Nuovo Pignone	2.0	25.0
KG2-3E	Dresser-Rand	2.1	16.9
M1A-23	KAWASAKI Heavy Industries	2.2	26.2
M1T-01	KAWASAKI Heavy Industries	2.3	20.4
M1A-13CC (steam injection)	KAWWASAKI Heavy Industries	2.4	33.7
CX501-KB3	Centrax Gas Turbine	2.7	25.0
SB15	MITSUI Eng. & S.B. Co.	2.7	25.6
M1T-03	KAWASAKI Heavy Industries	2.8	21.4
501-KB3	Allison Engine Company	2.8	26.0
M1T-13	KAWASAKI Heavy Industries	3.1	25.1
CX501-KN3	Cantrax Gas Turbine	3.1	27.1
TGC308CC	Tuma Turbomach	3.5	27.9
Centaur 40 & 40s	Solar	3.5	27.9
CX501-KB5	Centrax Gas Turbine	3.8	27.7
TB5000	European Gas Turbines	3.8	25.8
501/KB5	Rolls–Royce	3.9	29.3
AS4055	Allied Signal Engines	4.0	38.1
TB5000	European Gas Turbines	4.0	27.1
501-KH5S	Allison Engine Company	4.1	29.5
TGC378CH	Tuma Turbomach	4.1	28.0
M1T-23	KAWASAKI Heavy Industries	4.2	26.0
Typhoon	European Gas Turbines	4.2	29.9
Typhoon	Alstom[a]	4.3	N/A
501-KB4	Allison Engine Company	4.3	29.2
Centaur 50 & 50s	Solar	4.4	28.8
DR990	Dresser-Rand	4.4	30.5
CX501-KN5	Centrax Gas Turbine	4.5	29.9
Typhoon	Alstom[a]	4.7	N/A

(*continued on next page*)

Table 1 (*continued*)

Turbine	Manufacturer	Power (MWe)	Efficiency
TOC435CT	Tuma Turbomach	4.8	30.3
Typhoon	European Gas Turbines	4.9	33.2
Typhoon	European Gas Turbines	4.9	30.6
Typhoon	Stewart and Stevenson	4.9	30.6
Typhoon	Alstom[a]	5.0	N/A
CX501-KB7	Centrax Gas Turbine	5.0	29.3
Taurus 60 & 60s	Solar	5.0	30.3
Typhoon	Alstom[a]	5.2	N/A
501-KB7	Allison Engine Company	5.2	31.5
PGT5	Nuovo Pignone	5.2	26.9
501/KB7	Rolls–Royce	5.3	31.6
CX571	Centrax Gas Turbine	5.4	30.3
SB30	MITSUI Eng. & S.B. Co.	5.4	26.0
CX501-KN7	Centrax Gas Turbine	5.6	31.0
571-K	Allison Engine Company	5.9	33.9
MF-61	MITSUBISHI Heavy Industries	5.9	28.6
M7A-01	KAWASAKI Heavy Industries	6.0	30.5
CX501-KH	Centrax Gas Turbine	6.0	37.4
Tornado	European Gas Turbines	6.2	30.3
Tornado	Stewart and Stevenson	6.3	30.3
Taurus 70 & 70s	Solar	6.3	31.3
501/KH-5	Rolls–Royce	6.4	39.8
Tornado	European Gas Turbines	6.6	31.7
Tornado	Alstom[a]	6.7	N/A
501-KH (steam injection)	Allison Engine Company	6.8	39.9
Tempest	Stewart and Stevenson	7.5	31.4
Tempest	Alstom[a]	7.9	N/A
TG7	FIAT-TTG	8.6	24.2
Mars 90 & 90s	Solar	9.3	31.7
TGC880CM	Tuma Turbomach	9.3	31.8
G3142R(J)	European Gas Turbines	10.0	32.9
G3142R	Thomassen International	10.0	32.9
PGT10	Nuovo Pignone	10.1	30.9
G3142(J)	European Gas Turbines	10.4	25.6
M3142R	European Gas Turbines	10.4	34.4
G3142	Thomassen International	10.5	25.6
Mars 100 & 100s	Solar	10.7	32.5
TGC100CM	Tuma Turbomach	10.7	32.4
M3142	European Gas Turbines	10.9	26.7
SB60	MITSUI Eng. & S.D. Co.	12.5	29.6
MF-111A	MITSUBISHI Heavy Industries	12.6	30.3
Cyclone	Alstom[a]	12.9	N/A
PGT16	Nuovo Pignone	13.4	35.2
RLM1600	European Gas Turbines	13.4	35.4
LM1600PA	KVAERNER ENERGY AS	13.4	35.7
TG1600	Stewart and Stevenson	13.4	35.7
SB60	MITSUI Eng. & S.B. Co.	13.6	29.7
DR-60G	Dresser-Rand	13.6	35.9
LM1600PA	General Electric	13.8	35.5

(*continued on next page*)

Table 1 (*continued*)

Turbine	Manufacturer	Power (MWe)	Efficiency
RLM1600	European Gas Turbines	14.0	37.1
DR60G	Dresser-Rand	14.1	37.2
TGC111MF	Tuma Turbomach	14.3	33.0
MF-111B	MITSUBISHI Heavy Industries	14.6	31.0
Coberra 2000	Cooper Rolls, Inc.	14.6	28.2
LM1600 STIG (steam injection)	General Electric	17.0	39.5
GT35	Alstom[a]	17.0	32.1
OGT15000	Mashprom/Orenda	17.1	34.2
TG16	FIAT TTG	18.4	26.8
FT4A-9	Greenwich Turbine, Inc.	19.8	28.1
PG5271	Thomassen International	20.3	26.6
LM2500	Fiat Avio Power Division	21.9	35.5
RLM2500-PE	European Gas Turbines	21.9	35.6
PGT 25	Nuovo Pignone	21.9	35.5
DR-61	Dresser-Rand	22.1	36.1
LM2500 PE	KVAERNER ENERGY AS	22.2	36.3
LM2500	Nuovo Pignone	22.3	36.1
DR-61G	Dresser-Rand	22.8	36.8
RLM2500-PE	European Gas Turbines	22.8	36.8
LM2500PE	General Electric	22.8	36.8
LM2500	Greenwich Turbine, Inc.	22.8	36.8
TG2500	Stewart and Stevenson	22.8	36.8
DR61	Dresser-Rand	23.0	37.4
SB120	MITSUI Eng. & S.B. Co	23.0	30.5
DR61G	Dresser-Rand	23.3	37.6
RLM2500	European Gas Turbines	23.3	37.6
M5322R	European Gas Turbines	23.9	36.0
GT10B	Alstom[a]	24.8	34.2
FT8	Ebara Corporation	25.4	38.1
FT8	United Technologies	25.5	38.1
FT8	Ebara	26.1	39.2
PG 5371PA	European Gas Turbines	26.3	28.5
PG5371	John Brown Engineering	26.3	28.5
PG5371PA	KVAERNER ENERGY AS	26.3	28.5
MS5001	Nuovo Pignone	26.3	28.5
PG5371	Thomassen International	26.3	28.4
M5352	European Gas Turbines	26.6	36.4
MFT-8	MITSUBISHI Heavy Industries	26.8	38.7
RLM2500+	European Gas Turbines	27.0	36.6
DR-61G PLUS	Dresser-Rand	27.0	36.6
TG2500+	Stewart and Stevenson	27.1	36.6
Coberra 6000	Cooper Rolls, Inc.	27.2	35.8
RB211	Parsons Power Generation	27.2	35.8
RB211/6562	Rolls–Royce	27.5	36.2
DR61G PLUS	Dresser-Rand	27.6	37.3
RLM2500+	European Gas Turbines	27.6	37.3
DR-61 PLUS	Dresser-Rand	27.6	37.4
LM2500 STIG (steam injection)	General Electric	28.1	41.0
FT4C-3F	Greenwich Turbine, Inc.	28.1	30.7
M5382	European Gas Turbines	28.4	29.3

(*continued on next page*)

Table 1 (*continued*)

Turbine	Manufacturer	Power (MWe)	Efficiency
DR61 PLUS	Dresser-Rand	28.5	38.3
GT10C	Alstom[a]	29.1	36.0
RB211/6762	Rolls–Royce	29.5	37.7
MF-221	MITSUBISHI Heavy Industries	30.0	32.0
RB11/6761	Rolls–Royce	32.1	39.3
RLM5000-PC	European Gas Turbines	34.3	36.8
RLM5000-PC	European Gas Turbines	34.3	36.5
TG5000	Stewart and Stevenson	34.4	37.2
LM5000PC	General Electric	34.5	37.2
RLM5000	European Gas Turbines	35.1	37.8
MW-251	MITSUBISHI Heavy Industries	36.8	28.9
PG 6541B	European Gas Turbines	38.3	31.4
PG6541B	Greenwich Turbine, Inc.	38.3	31.4
PG6541	John Brown Engineering	38.3	31.4
PG6541B	KVAERNER ENERGY AS	38.3	31.4
MS6001	Nuovo Pignone	38.3	31.4
PG6541	Thomassen International	38.3	31.4
TG20	FIAT TTG	38.4	30.7
TG20B7/8U	Fiat Avio Power Division	39.4	29.9
LM6000PA	KVAERNER ENERGY AS	39.6	39.7
LM6000	Greenwich Turbine, Inc.	39.9	38.8
LM6000	John Brown Engineering	40.0	38.8
LM6000	Nuovo Pignone	40.0	38.9
LM6000	Fiat Avio Power Division	40.5	39.1
RLM6000	European Gas Turbines	40.6	39.5
DR-63G	Dresser-Rand	40.7	39.2
TG 6000	Stewart and Stevenson	40.8	39.7
DR63G	Dresser-Rand	42.0	40.5
MS6001B	General Electric	42.1	32.1
MS6001C	General Electric	42.3	36.3
GTX 100	Alstom[a]	43.0	37.0
LM6000PC	General Electric	43.9	41.9
TG20B11/12	Fiat Avio Power Division	47.8	33.5
251B11	Parsons Power Generation	49.2	32.7
FT 8 Twin	Ebara Corporation	51.1	38.3
TRENT	Parsons Power Generation	51.2	41.6
Trent	United Technologies	51.2	41.6
LM5000 STIG (steam injection)	General Electric	51.6	43.8
Trent 50	Rolls–Royce	51.9	42.2
GT8C2	Alstom[a]	57.0	34.0
Trent 60	Rolls–Royce	58.2	40.8
V64.3	Ansaldo Energia	63.0	35.4
V64.3A	Siemens/Westinghouse	67.0	N/A
PG6101	John Brown Engineering	70.1	34.2
PG6101FA	KVAERNER ENERGY AS	70.1	34.2
MS6001FA	Nuovo Pignone	70.1	34.2
PG6101	Thomassen International	70.1	34.3
PG6101FA	General Electric	70.2	34.2
MS6001FA	General Electric	75.9	35.0

(*continued on next page*)

Table 1 (*continued*)

Turbine	Manufacturer	Power (MWe)	Efficiency
PG7111	John Brown Engineering	83.5	32.6
PG7111EA	KVAERNER ENERGY AS	83.5	32.6
MS7001E	Nuovo Pignone	83.5	32.6
TG50	FIAT TTG	92.7	31.2
MW-501	MITSUBISHI Heavy Industries	104.5	33.3
V84.2	Ansaldo Energia	109.0	33.7
GT11N2	Alstom[a]	114.7	33.4
PG 9171E	European Gas Turbines	123.4	33.8
PG9171	John Brown Engineering	123.4	33.8
PG9171E	KVAERNER ENERGY AS	123.4	33.8
MS9001E	Nuovo Pignone	123.4	33.8
PG9171	Thomassen International	123.4	33.7
MS9001E	General Electric	126.1	33.8
MW-701	MITSUBISHI Heavy Industries	130.5	33.9
MW-701DA	MITSUBISHI Heavy Industries	136.9	34.0
701DA	Parsons Power Generation	138.3	34.2
TG50D5	Fiat Avio Power Division	140.8	34.5
TG50D5S	Fiat Avio Power Division	147.8	34.5
V84.3	Ansaldo Energia	154.0	36.2
V94.2	Siemens/Westinghouse	157.0	34.3
501F	MITSUBISHI Heavy Industries	158.6	36.0
V94.2	Ansaldo Energia	159.0	34.2
PG7221FA	KVAERNER ENERGY AS	159.0	35.9
GT13E2	Alstom[a]	165.0	35.7
PG7231	John Brown Engineering	167.8	36.2
PG9231	Thomassen International	168.9	34.9
MS9001EC	Nuovo Pignone	169.2	35.0
GT24	Alstom[a]	171.0	36.5
V94.2A	Siemens/Westinghouse	190.0	35.2
V94.3	Ansaldo Energia	222.0	36.2
PG9311FA	General Electric	226.0	35.7
PG9331 FA	European Gas Turbines	226.5	35.7
PG9331	John Brown Engineering	226.5	35.7
PG9331 FA	KVAERNER ENERGY AS	226.5	35.7
MS901FA	Nuovo Pignone	226.5	35.7
PG9331	Thomassen International	226.5	35.6
501G	MITSUBISHI Heavy Industries	230.0	38.5
701F	Fiat Avio Power Division	234.0	36.7
701F	MITSUBISHI Heavy Industries	234.2	36.6
701F	Parsons Power Generation	236.7	36.8
M89001FA	General Electric	255.6	36.9
V94.3A	Siemens/Westinghouse	265.0	38.5
GT26	Alstom[a]	268.0	37.0

[a] Gas turbine business of Alstom has been recently sold to Siemens.

temperature versus entropy (*T–S*) diagrams for this cycle. The numbers on this diagram correspond to the numbers also used in Fig. 1. Path 1 to 2 represents the compression occurring in the compressor, path 2 to 3 represents the constant pressure addition of heat in the combustion chamber, and path 3 to 4 represents the expansion occurring in the turbine.

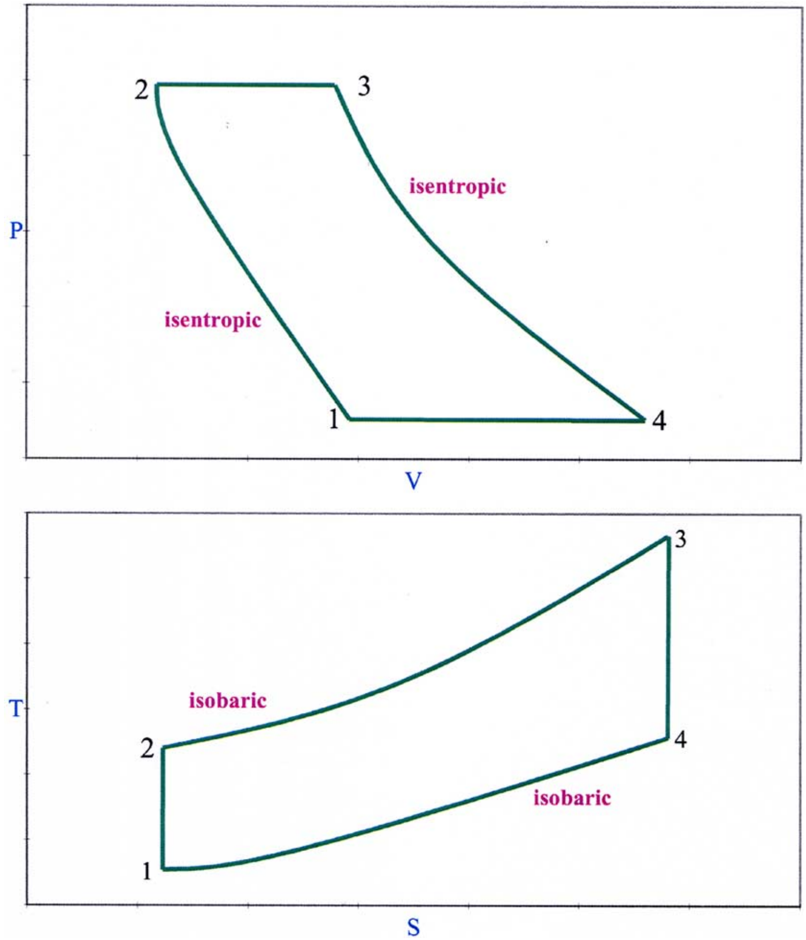

Fig. 2. The Brayton cycle.

The path from 4 back to 1 on the Brayton cycle diagrams indicates a constant pressure cooling process. In the open cycle gas turbine, this cooling is done by the atmosphere, which provides fresh, cool air at point 1 on a continuous basis in exchange for the hot gases exhausted to the atmosphere at point 4.

The Brayton cycle is characterised by two significant parameters, i.e. (a) pressure ratio and (b) firing temperature. The pressure ratio of the cycle is the pressure at point 2 (compressor discharge pressure) divided by the pressure at point 1 (compressor inlet pressure). In an ideal cycle, this pressure ratio is also equal to the pressure at point 3 divided by the pressure at point 4. However, in an actual cycle there is some slight pressure loss in the combustion chamber and, hence, the pressure at point 3 is slightly less than at point 2. The other significant parameter, the firing temperature, is defined as the highest temperature reached in the cycle. High-pressure ratio results in power output increase

and maximum efficiency change with firing temperature increase. The higher the pressure ratio, the greater the benefits from increased firing temperature.

2.3. Factors affecting gas turbine performance

The performance of gas turbine is affected by various factors such as fuel type, inlet air properties, water injection or steam injection, etc. The effect of these factors is briefly discussed in the following sections.

2.3.1. Fuel

Work from a gas turbine is defined as the product of mass flow, heat energy in the combusted gas and temperature differential across the turbine. The mass flow is the sum of compressor airflow and fuel flow. The heat energy is a function of the elements in the fuel and the products of combustion. Natural gas produces approximately 2% more output than does gasoil fuel. This is due to the higher specific heat in the combustion products of natural gas, resulting from the higher water vapour content produced by the higher hydrogen/carbon ratio of natural gas. This effect is noted even though the mass flow of natural gas is lower than the mass flow of gasoil fuel, due to the greater net calorific value of the natural gas.

2.3.2. Fuel heating

Most of the combined cycle power plants are designed for maximum efficiency. These plants often utilise integrated fuel gas heaters. Heated fuel results in higher gas turbine efficiency due to the reduced fuel flow required to raise the total gas temperature to firing temperature. Fuel heating will result in slightly lower gas turbine output because of the incremental volume flow decrease. Typically, the source of heat for the fuel is the steam cycle feed water. Since use of this energy in the gas turbine fuel heating system is thermodynamically advantageous, the combined cycle efficiency can be improved by approximately 0.6%.

2.3.3. Air temperature and site elevation

The gas turbine performance is changed by anything that affects the density or the mass flow of the air intake to the compressor. Ambient weather conditions are the most obvious changes from the ISO reference conditions of 15 °C and 1013 bar. Fig. 3 shows how ambient temperature affects the power output and the heat rate of a typical simple-cycle gas turbine. Each gas turbine model has its own ambient temperature effect cure, as it depends on the cycle parameters and component efficiencies as will as air mass flow.

Correction for altitude or barometric pressure is more straightforward. The air density reduces as the site elevation increases. While the resulting airflow and output decrease proportionately, the hate rate and other cycle parameters are not affected. A standard altitude correction curve is presented in Fig. 4.

2.3.4. Inlet air cooling

The ambient effect curve, presented in Fig. 3, clearly shows that the turbine output and the heat rate are improved as compressor inlet temperature decreases. Lowering the compressor, inlet temperature can be accomplished by installing an evaporative cooler or

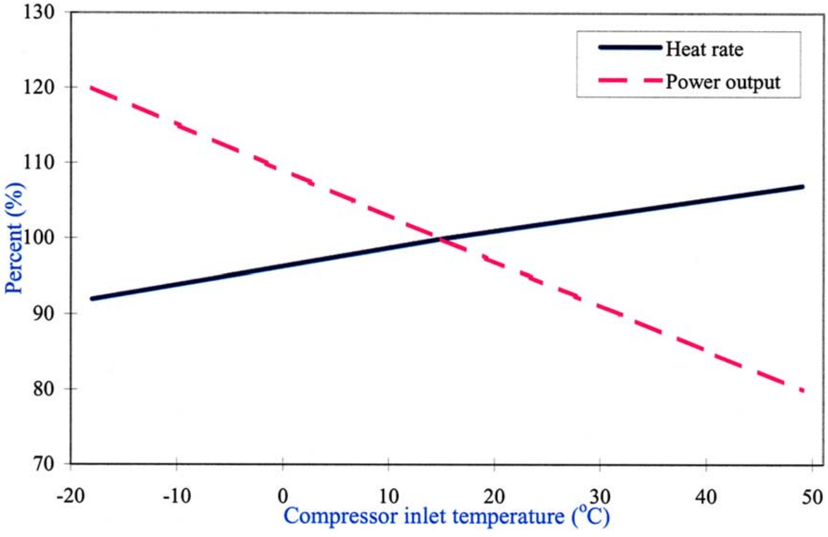

Fig. 3. Effect of ambient temperature.

a chiller in the inlet ducting downstream of the inlet filters [2]. Careful application of these systems is necessary, as condensation or carryover of water can results to compressor fouling and degrade performance, These systems generally are followed by moisture separators or coalescing pads to reduce the possibility of moisture carryover. The biggest gains from evaporative cooling are realized in hot, low-humidity climates [25,38].

2.3.5. Humidity

Humid air, which is less dense than dry air, also affects output and heat rate, as shown in Fig. 5. In the past, this effect was thought to be too small to be considered. However, with

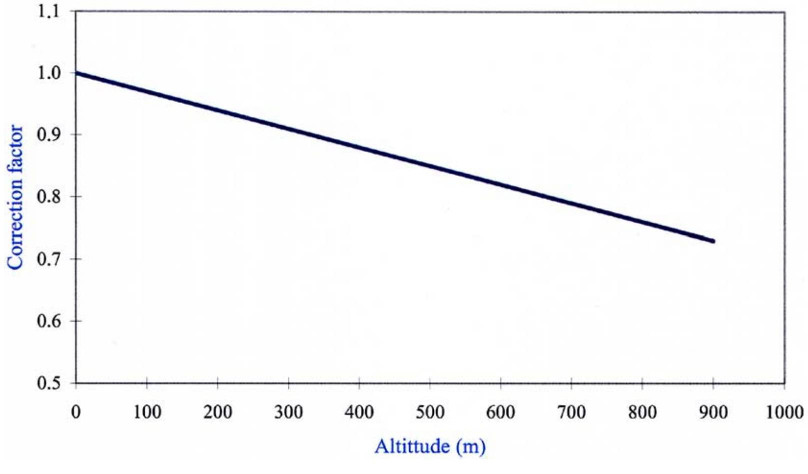

Fig. 4. Altitude correction curve.

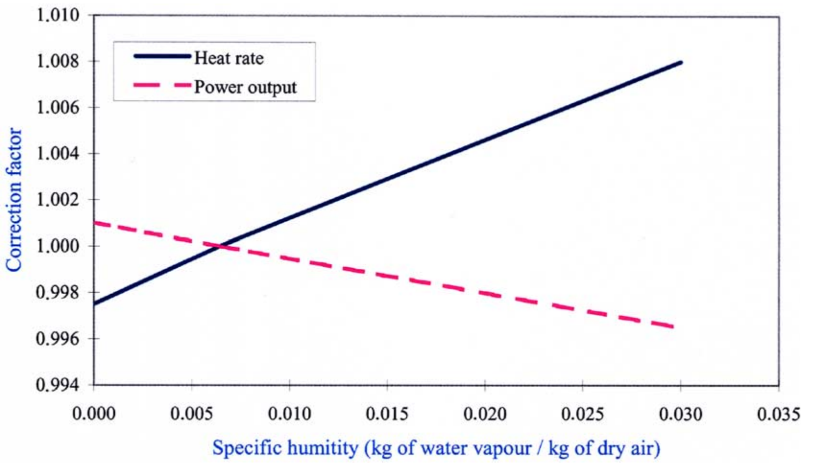

Fig. 5. Humidity effect curve.

the increasing size of gas turbines and the utilization of water and steam injection for NO_X emissions control, this effect has greater significance.

2.3.6. Inlet and exhaust losses

Inserting air filtration, silencing, evaporative coolers or chillers into the inlet or heat recovery devices in the exhaust causes pressure losses in the system. The effects of these pressure losses are unique to each design. Typically, with a 10 mbar pressure drop at the air inlet to the compressor (a) the heat rate can be increased by 0.4%, (b) the power output can be decreased by 1.4% and (c) the exhaust temperature can be increased by 1 °C. Similarly, with a 10 mbar pressure drop at the exhaust gas outlet (a) the heat rate can be increased by 0.4%, (b) the power output can be decreased by 0.4% and (c) the exhaust temperature can be increased by 1 °C.

2.3.7. Air extraction

In some gas turbine applications, it may be desirable to extract air from the compressor. Generally, up to 5% of the compressor airflow can be extracted from the compressor discharge casing without modification to casings or on-base piping. Pressure and air temperature will depend on the type of machine and site conditions, Air extraction between 6 and 20% may be possible, depending on the machine and combustion chamber configuration, with some modifications to the casings, piping and controls. Air extractions above 20% will require extensive modification to the turbine casing and unit configuration. As a rule of thumb, every 1% in air extraction results in a 2% loss in power.

2.3.8. Performance degradation

All turbomachinery experiences losses in performance with time. Gas turbine performance degradation can be classified as recoverable or non-recoverable loss. Recoverable loss is usually associated with compressor fouling and can be partially rectified by water washing or, more thoroughly, by mechanically cleaning the compressor

blades and vanes after opening the unit. Non-recoverable loss is due primarily to increased turbine and compressor clearances and changes in surface finish and airfoil contour. Because this loss is caused by reduction in component efficiencies, it cannot be recovered by operational procedures, external maintenance or compressor cleaning, but only through replacement of affected parts at recommended inspection intervals. Recent field experience indicates that frequent off-line water washing is not only effective in reducing recoverable loss, but also reduces the rate of non-recoverable loss. One generalization that can be made from the data is that machines located in dry, hot climates typically degrade less than those in humid climates.

2.3.9. Diluent injection

Water or steam injection for NO_X emissions control to meet applicable regulations is commonly used. This is accomplished by admitting water or steam in the combustion chamber. Each gas turbine configuration has limits on water or steam injection levels to protect the combustion system and turbine section. Depending on the amount of water or steam injection needed to achieve the desired NO_X level, output will increase because of the additional mass flow. Fig. 6 shows the effect of steam injection on the power output for a typical gas turbine.

2.4. Steam and water injection for power augmentation

As mentioned in Section 2.3 by injecting steam or water into combustion chamber for NO_X emissions abatement increases mass flow and, therefore, power output. Generally, the amount of water is limited to the amount required to meet the NO_X requirement in order to minimise operating cost and impact on inspection intervals. When steam is injected for power augmentation, it can be introduced into compressor discharge casing of the gas turbine as well as into the combustion chamber [24]. The effect on output and heat rate is the same as that shown in Fig. 6. Typically, gas turbines are designed to allow up to 5% of

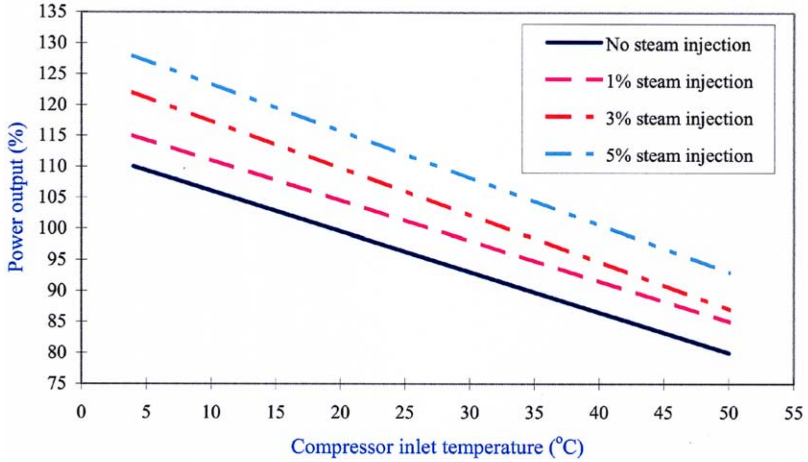

Fig. 6. Effect of steam injection on power output.

the compressor airflow for steam injection to the combustion chamber and compressor discharge. When either steam or water is used for power augmentation, the control system is normally designed to allow only the amount needed for NO_X abatement until the machine reaches full load. At that point, additional steam or water can be admitted via the governor control. The various available MAST technologies are presented in Section 4.

3. Non-MAST technologies

3.1. The gas to gas recuperation cycle

Gas turbine efficiency can be raised when gas to gas recuperation is employed and this has been used in conjunction with industrial gas turbines for more than 50 years. This arrangement is illustrated in Fig. 7. The use of recuperation is limited, however, by the compressor outlet temperature due to metallurgical problems of the heat exchanger temperature. Inter-cooling reduces the heat transfer problem and allows recuperation with high efficiency turbines.

This concept is used in several gas turbines, such as the 1.4 MWe Heron gas turbine [16], the 21 MWe Rolls–Royce WR-21 gas turbine [7,21], or the Solar gas turbines in the 1–25 MWe size range [12]. The recuperated gas turbines are expected to obtain efficiencies from 39 to 43%, which are higher compared to 25–40% for other simple-cycle gas turbines of same capacity.

There is a view, which is well supported theoretically, that a regenerator would provide a more efficient cycle than a recuperator. This is due to the very high thermal effectiveness of regenerators since values of around 95% are possible. However, maximum efficiency occurs at very low-pressure ratios, typically less than 5. Consequently, even if the theoretical performance could be achieved, relatively large turbines would be required [10].

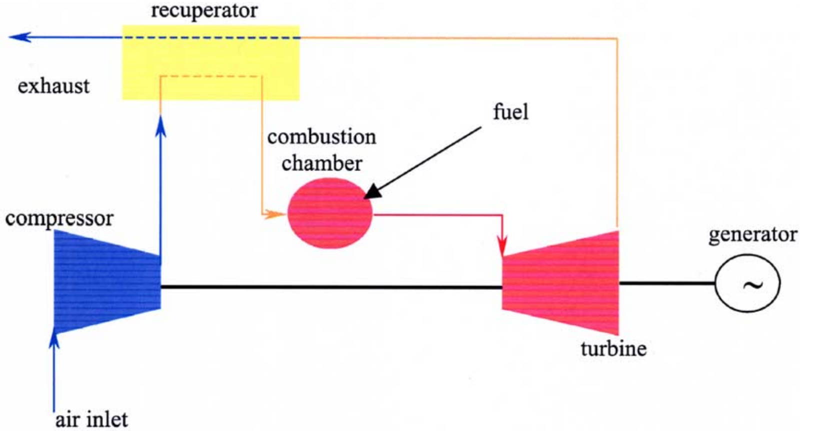

Fig. 7. Gas to gas recuperation.

3.2. The combined cycle

A typical simple-cycle gas turbine will convert 30–40% of the fuel input into shaft output. All but 1–2% of the remainder is in the form of exhaust heat. The Brayton–Rankine cycle, commonly referred as to the conventional combined cycle is the well-known arrangement of a gas turbine with a steam turbine bottoming cycle. The combined cycle is generally defined as one or more gas turbines with heat recovery steam turbines in the exhaust, producing steam for a steam turbine generator. Fig. 8 shows a combined cycle in its simplest form. High utilization of the fuel input to the gas turbine can be achieved with some of the more complex heat recovery cycle, involving multiple-pressure boilers, extraction or topping steam turbines, and avoidance of steam flow to a condenser to preserve the latent heat content. Attaining more than 80% utilization of the fuel input by a combination of electrical power generation and process heat is not unusual. Combined cycles producing only electrical power are in the 50–58% thermal efficiency range using the more advanced gas turbines.

In a typical scheme, shown in Fig. 8, exhaust heat from the open gas turbine circuit is recovered in a heat recovery steam generator. In order to provide better heat recovery in the heat recovery steam generator, more than one pressure level is used. With a single pressure heat, recovery steam generator typically about 30% of the total plant output is generated in the steam turbine. A dual pressure arrangement can increase the power output of the steam cycle by up to 10%, and an additional 3% can result by choosing a triple pressure cycle [30]. Modem gas turbine combined cycle plants with a triple pressure heat recovery steam generator with steam reheat can reach efficiencies above 55%. Siemens/ Westinghouse claims 58% efficiency [22], Alstom claims 58.5% efficiency [20]

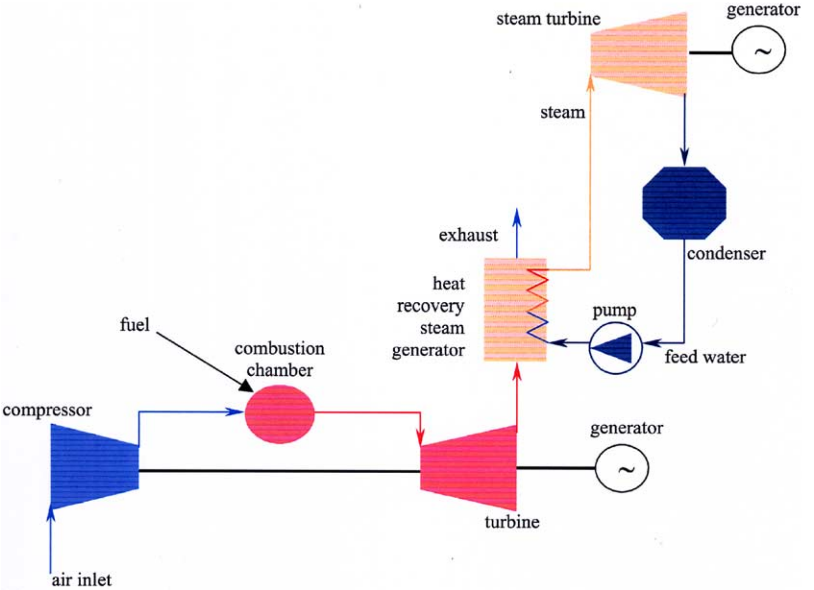

Fig. 8. The combined cycle.

and General Electric claims an efficiency of 60% [15]. These high efficiency values can be achieved at large units above 300 MWe.

Combined cycle plants have become a well-known and substantial technology for power generation due to its numerous advantages including high efficiency and low emissions. The combined cycle technology provides a range of advantages [32].

These include:

(a) Higher thermal efficiency from any other gas turbine advance cycle. The best conventional oil- and coal-fired power plants on the market have thermal efficiencies around 43–45%.
(b) Low emissions since natural gas produce no ash or SO_X, less quantities of volatile hydrocarbons, carbon monoxide and NO_X than oil and coal and produces much less CO_2.
(c) Low capital costs and short construction times (often 2–3 years).
(d) Less space requirements than the space required for equivalent coal or nuclear stations, which reduces site constraints.
(e) Flexibility in plant size with maximum power outputs range between 10 and 750 MWe per combined cycle-unit.
(f) Fast start-up which makes it easier to respond to change in demand.

3.3. The Brayton–Kalina cycle

The Kalina cycle is a novel bottoming cycle, which uses zeotropic mixture of ammonia and water as the working fluid. Its characteristics are such that its temperature tracks the turbine exhaust temperature in the waste heat boiler. However, the thermodynamic advantage of this small boiler temperature difference compared to a steam cycle would be lost at the condensing stage, assuming the condenser cooling medium temperature would be the same in both cases. The novelty of the Kalina cycle lies in the solution to this problem.

The principle of the Brayton–Kalina cycle is illustrated schematically in Fig. 9. There are, in effect, two condensing stages. In the first stage, the turbine exhaust steam is fully absorbed on a continuous basis, by a stream of secondary fluid in a liquid phase and the heat of absorption is dissipated into the condenser cooling water. The secondary fluid is an ammonia/water mixture of different composition from the turbine exit stream but this is not important with respect to the underlying principle. Following the absorption process, the mixture of secondary fluid plus turbine exit stream is pressurised by a pump. Since a liquid is being compressed, very little pumping work is required. The pressurised fluid is heated by the turbine exhaust stream and this cause the turbine working fluid to boil off from the secondary fluid. Since the fluid is now at a higher pressure than at the turbine exit it can be fully condensed by the condenser cooling water.

The Kalina cycle can produce 10–30% more power than a Rankine cycle [28]. Because the exhaust pressure of the vapour turbine in the Kalina cycle is above atmospheric pressure, no vacuum is needed to be maintained in the condenser during operation, or stand-by periods. Therefore, the start-up procedure can be performed in a much shorter time. The working fluid composition can easily be changed in order to obtain the optimal performance with respect to alterations in load or ambient conditions. Another advantage

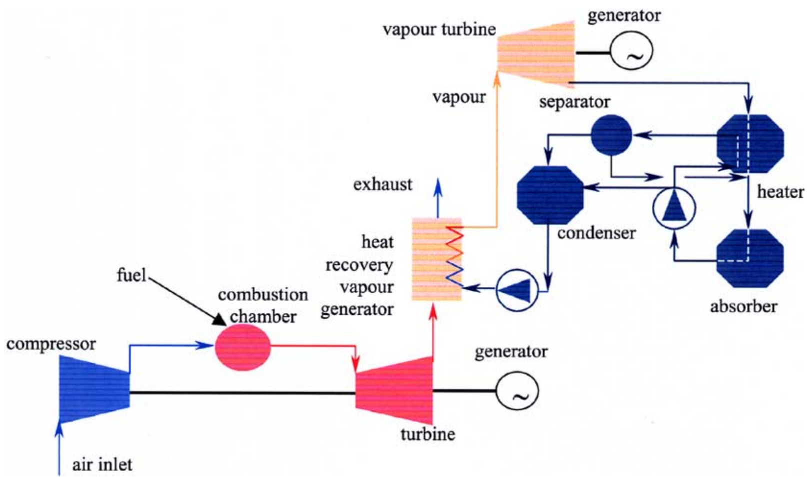

Fig. 9. The Brayton–Kalina cycle.

is the smaller size of the whole unit. The footprint of the Kalina plant is about 60% of the size of a Rankine plant design.

In 1993, General Electric signed an exclusive worldwide licensing agreement with the Kalina cycle patent owner (Exergy, Inc.) to design and market Beayton–Kalina cycle plants. The General Electric commercial size demonstration plant, of 260 MW capacity, was planned to be in operation by 1998, however, the project was suspended [17].

3.4. The Brayton–Brayton cycle

Two Brayton cycles can be combined by an air–gas heat exchanger as illustrated in Fig. 10. The exhaust of the primary gas turbine is sent to a heat exchanger, which, in turn, heats the air in the secondary gas turbine cycle. Air is expanded in the turbine to generate additional power. In comparison to the conventional combined cycle, this scheme does not require bulky steam equipment (boiler, steam turbine, condenser), or a water processing unit, and allows unmanned operation.

Recent studies showed the feasibility of this configuration [28]. These reported an increase of power by 18–30% depending on the number of intercoolers, and an efficiency growth of up to 10% points. For example, for the Allison 571 K topping gas turbine [21], introduction of the air bottoming cycle with two intercoolers led to an increase in power from 5.9 to 7.5 MWe and in efficiency from 33.9 to 43.2%. Comparable results were obtained with the General Electric LM2500 topping turbine [15]. The scheme can also be applied for cogeneration. The exhaust air, leaving the cycle at 200–250 °C, can be used for process needs that require heat of such temperatures.

3.5. The Brayton–Diesel cycle

Preheating of the inlet air of a Diesel engine can sufficiently improve its performance. The gas turbine exhaust can be applied in order to increase the temperature of the air, which

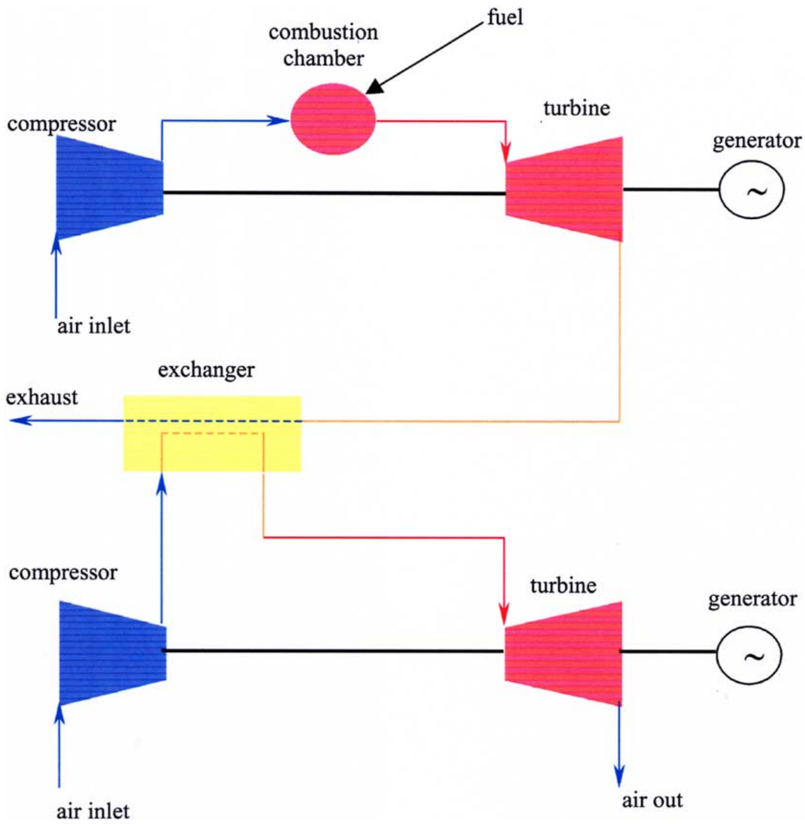

Fig. 10. The Brayton–Brayton cycle.

is extracted from the compressor and fed into the Diesel engine. Subsequently, the engine outlet flow expands through the low-pressure stage of the gas turbine as illustrated in Fig. 11.

3.6. The Brayton–Stirling cycle

In a combination of a gas turbine and a Stirling engine, the heater of the latter can be placed either in the combustion chamber of the turbine, or after the turbine in the exhaust flow, as shown in Fig. 12. The arrangement is determined by the optimal performance of the cycle and by the materials used in the Stirling heater's head. As much as 9 MWe can be recovered by a bottoming Stirling cycle from the exhaust of a Rolls–Royce RB211 gas turbine of 27.5 MWe [21]. Such a plant can obtain an efficiency of 47.7%. Just as the Brayton–Brayton cycle, this combination provides a compact and simple heat recovery scheme.

3.7. The Brayton–fuel cell cycle

A fuel cell system which offers high efficiency (60%), can operate at high pressure and can produce very high temperature exhaust gases, which allows integrating a gas turbine

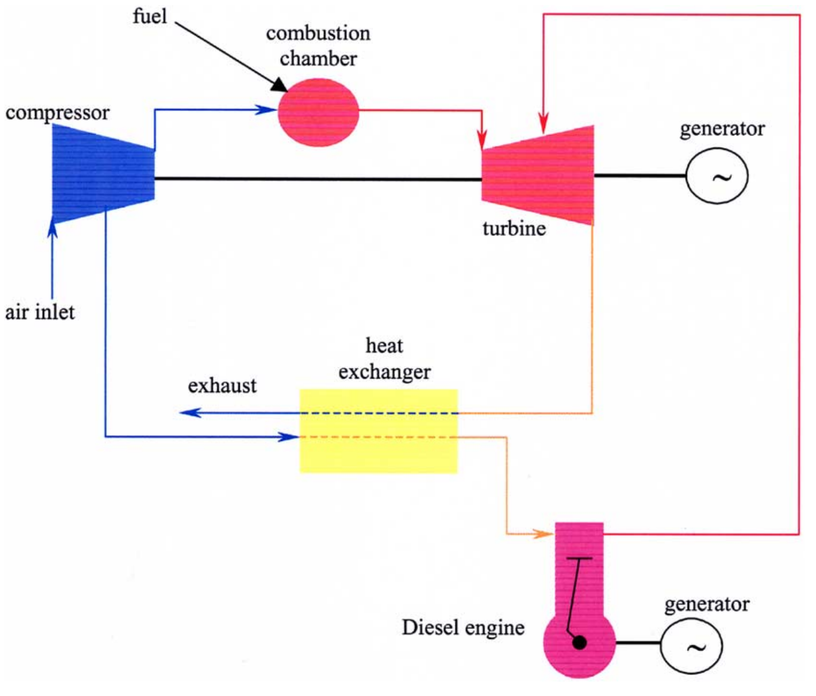

Fig. 11. The Brayton–Diesel cycle.

within the system, thus improving performance [13]. The schematic of the system is presented in Fig. 13. The use of the fuel cells integrated with combustion chambers allows efficiency to approach 70% [34]. The Brayton–fuel cell cycle is claimed to have the highest efficiency of any advance cycle, and can, therefore, be seen as a choice for the future power plants [23].

3.8. The chemical recuperation cycle

The chemical recuperation cycle uses a reforming process to convert methane, water, and sometimes CO_2 into a hydrogen and carbon monoxide fuel mixture that can be burned in the combustor. This endothermic reaction absorbs heat at a temperature lower than the combustion temperature and in this manner increases the fuel's heating value. Recuperation that proceeds thermally and chemically results in a higher degree of heat recovery than in standard recuperation schemes. Moreover, the hydrogen rich fuel has greater flammability than methane and supports combustion at a lower flame temperature, which potentially reduces NO_X formation. However, the gas turbine exhaust temperature is not high enough for a complete reforming reaction. At 550 °C, only 20% of the fuel is reformed. In order to increase the temperature, some additional firing can be applied. Different reforming schemes have been proposed [10].

In the scheme with steam reforming, steam generated in a heat recovery steam generator is mixed with natural gas in a reformer as illustrated in Fig. 14. A power plant

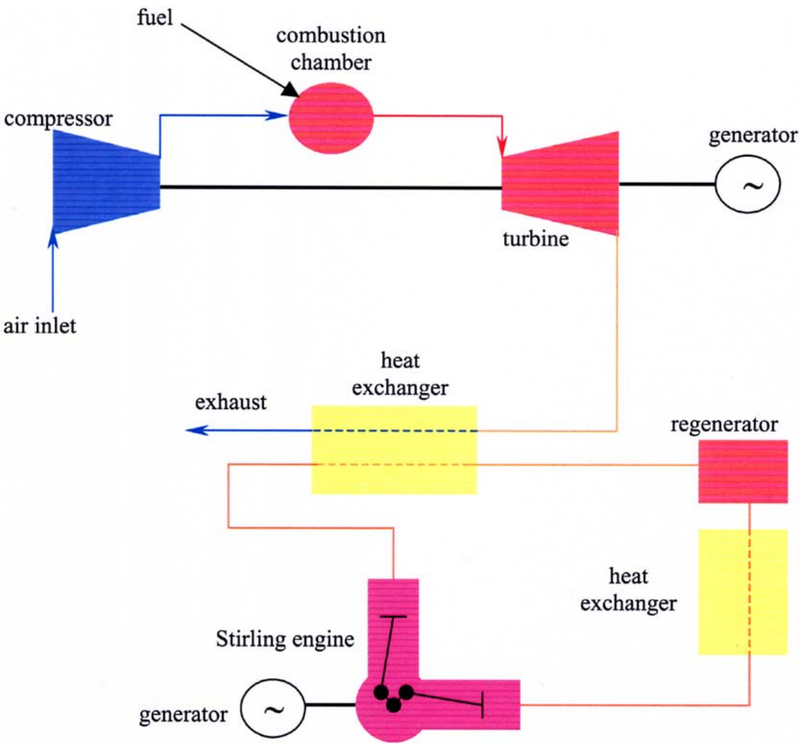

Fig. 12. The Brayton–Stirling cycle.

based on the General Electric LM5000PC gas turbine could reach an efficiency of 45% in comparison to 37.2% in simple-cycle [26]. Another scheme uses exhaust gas recuperation, a portion of the flue gas is compressed, mixed with natural gas, heated with exhaust heat from the combustion turbine, mixed with the air from the compressor, and sent to the combustion chamber (Fig. 15). When the mixture is heated in the presence of a nickel-based catalyst, hydrogen and carbon monoxide are produced. The reaction is accelerated at low excess oxygen, low pressure, and high mass ratio of recycled exhaust gas to methane. Therefore, the best results are achieved when the fuel is burned at the stoichiometric ratio, and the exhaust gas is used to reduce the turbine inlet temperature. Typical value of recirculation as over 50% of turbine flow. This means that both the air compressor flow and the exhaust flow are less than half that of conventional cycles with the same turbine size.

3.9. Other advance gas turbine cycles

Other advance gas turbine cycles which are currently under development utilising various cycle modifications involve the gas turbine turbocharged by the steam turbine (GTTST) cycle [29], the coal fire air turbine (CAT) cycle [19], the Gratz cycle [8], the chemical looping combustion (CLC) cycle [1] and the hydrogen combustion turbine [11].

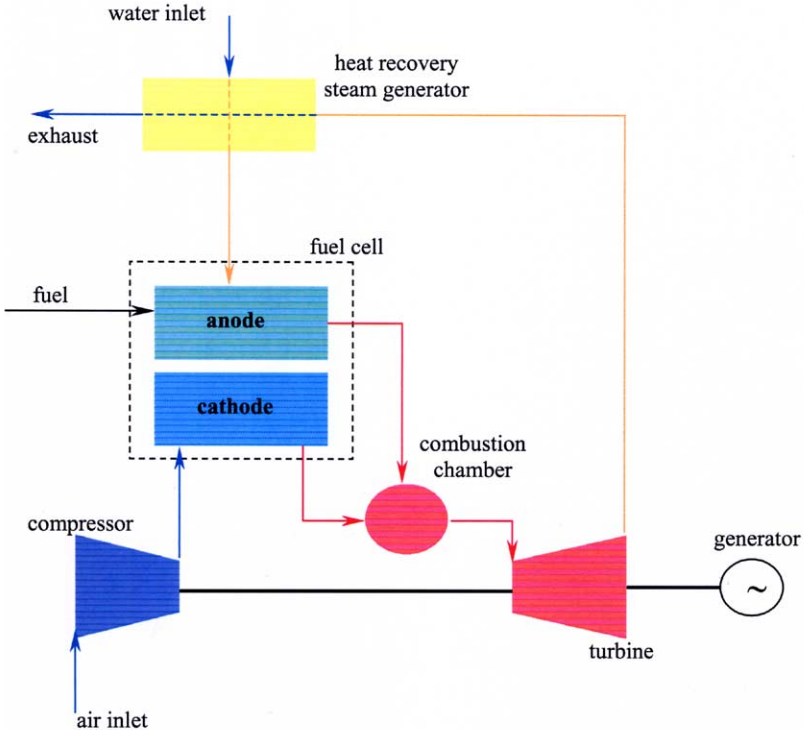

Fig. 13. The Brayton–fuel cell cycle.

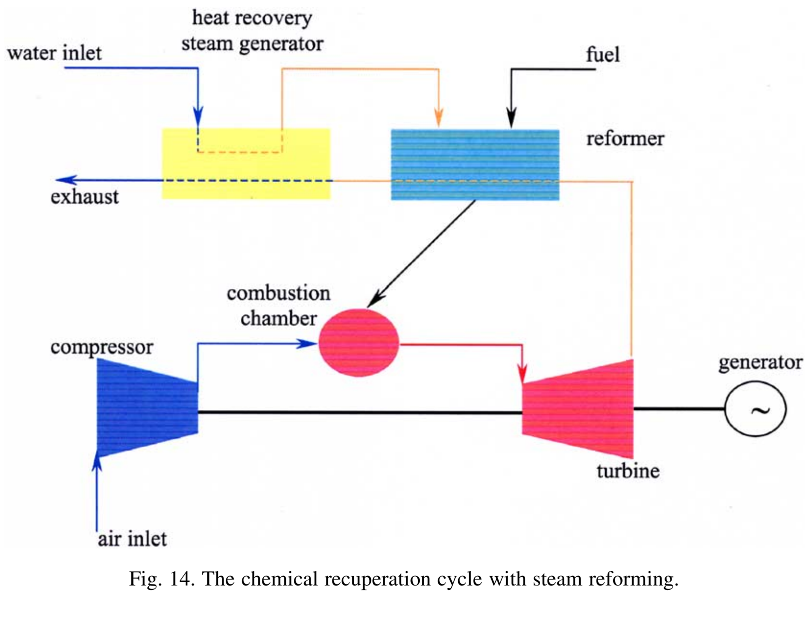

Fig. 14. The chemical recuperation cycle with steam reforming.

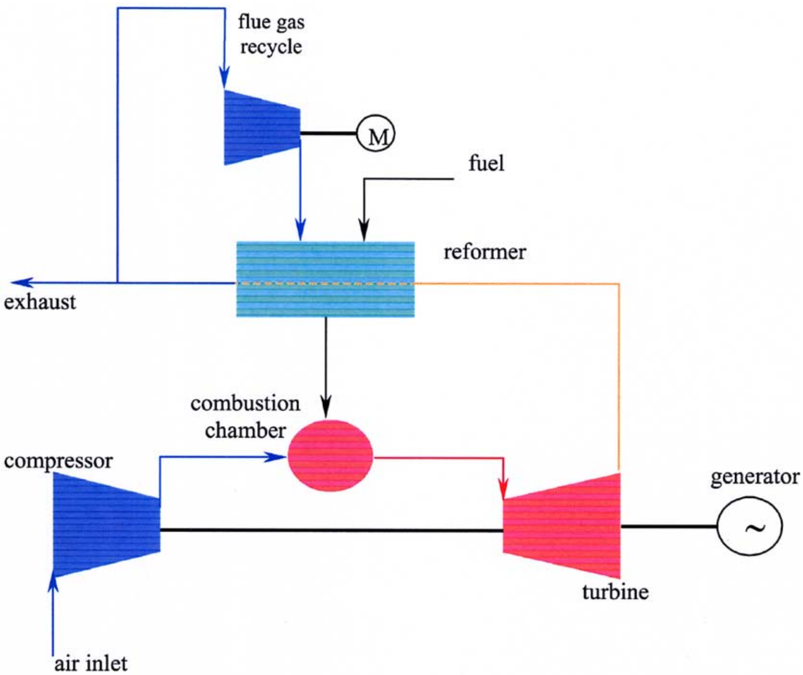

Fig. 15. The chemical recuperation cycle with flue gas recycling.

4. MAST technologies

The positive effect of the steam or water injection on the performance of a gas turbine has been briefly discussed in Section 2.4. Water injection has been used for power augmentation in aircraft engines since the 1950s, and in industrial gas turbines since 1960s. The injection increases the mass flow and the specific heat of the working fluid, which gives additional power to the cycle. Along with this, it helps to lower NO_X formation in the combustion chamber and to cool the blades more effectively than air.

Depending on the amount of water or steam injection output will increase because of the additional mass flow. Fig. 6 shows the effect of steam injection on the power output for a typical gas turbine.

As shown in Tables 2–4, there are five gas turbines on the market, which are adapted to the use of steam injection. The three bigger machines are all constructed by General Electric (General Electric dubbed its steam injected gas turbines STIG™ (STeam Injected Gas Turbine) [15]) and are the LM5000 STIG™, the LM2500 STIG™ and the LM1600 STIG™, producing, respectively, 51.6, 28.1 and 17 MWe. Without steam injection they produce 34.5, 22.8 and 13 MWe. The two smaller machines are the Allison 501-KH [21] and the Kawasaki M1A-13CC [18]. The most recent variant of the Allison 501 produces 4.9 MWe without steam injection and 6.8 MWe with steam injection. The latest development in steam injected gas turbines is the Kawasaki M1A-13CC. With this machine, Kawasaki aims at the low power cogeneration applications. The gas turbine

Table 2
Power output of the commercial available steam injection gas turbines

Turbine	Manufacturer	Power (MWe)	
		Without steam injection	With steam injection
M1A-13CC	KAWASAKI Heavy Industries	1.3	2.4
501-KH	Allison Engine Company	4.9	6 8
LM1600 STIG	General Electric	13.0	17.0
LM2500 STIG	General Electric	22.8	28.1
LM5000 STIG	General Electric	34.5	51.6

produces 2.4 MWe in steam injection mode and 1.3 MWe without steam injection. Various types of MAST turbines are currently under development; e.g. see [36].

In the following sections, an overview of the MAST technologies is presented. These include the Cheng cycle, the DRIASI cycle, the HAT cycle, the LOTHECO cycle, etc. The main characteristics are given along with a discussion on advantages and disadvantages of these schemes.

4.1. The Cheng cycle

In 1978, Cheng [4] proposed a gas turbine cycle in which the heat of the exhaust gas of the gas turbine is used to produce steam in a heat recovery steam generator as shown in Fig. 16. This steam is injected in the combustion chamber of the gas turbine, resulting in an efficiency gain and a power augmentation. The cycle is commonly called the Cheng cycle

Table 3
Efficiency of the commercial available steam injection gas turbines

Turbine	Manufacturer	Efficiency (%)	
		Without steam injection	With steam injection
M1A-13CC	KAWASAKI Heavy Industries	22.3	33.7
501-KH	Allison Engine Company	31.5	39.9
LM1600 STIG	General Electric	35.5	39.5
LM2500 STIG	General Electric	36.8	41.0
LM5000 STIG	General Electric	37.2	43.8

Table 4
Capital cost of the commercial available steam injection gas turbines [14]

Turbine	Manufacturer	Capital cost (€ kW)
MIA-13CC	KAWASAKI Heavy Industries	N/A
501-KH	Allison Engine Company	562
LM1600 STIG	General Electric	514
LM2500 STIG	General Electric	427
LM5000 STIG	General Electric	410

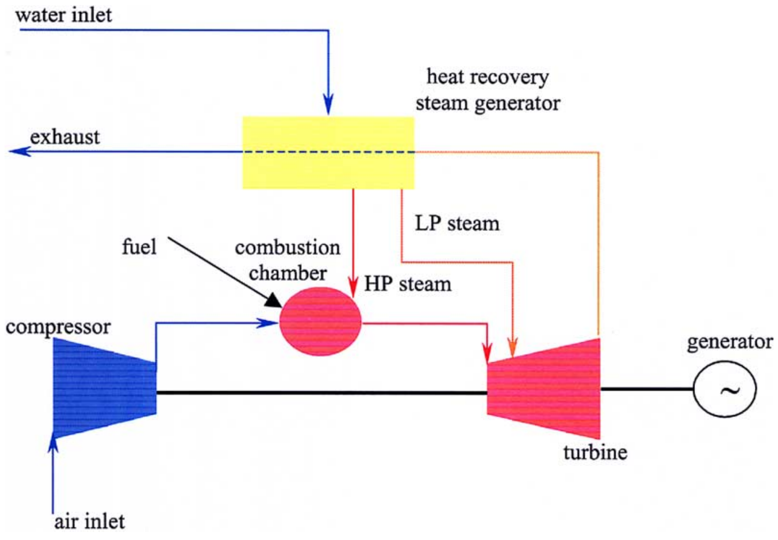

Fig. 16. The Cheng cycle.

or the steam injection cycle. High-pressure steam can be injected into the combustion chamber, while intermediate-pressure and low-pressure steam are often expanded in the first gas turbine stages, as shown in Fig. 16. The system will work if the pressure of the steam is higher than that at the compressor outlet.

By introducing steam injection in a gas turbine, an efficiency gain of about 10% and a power augmentation of about 50–70% are possible. Using a steam turbine to expand the steam, i.e. applying a conventional combined cycle instead of a Cheng cycle, gives higher efficiency gains. Accepted efficiency for the combined cycle is nowadays 50–58%, with a power rise of about 30–50% with respect to the simple cycle.

Expansion of steam in the gas turbine proceeds to the atmospheric pressure and in a less efficient manner than in the steam turbine. Whereas in the combined cycle plant steam leaves the steam turbine at much lower pressures, thus providing more power. Therefore, a gas turbine with steam injection will always have a lower efficiency than that in conventional combine cycle operation. The introduction of inter-cooling and reheat in a steam injection gas turbine allows to reduce the power consumed by the compressor from 50% of total output for modern engines down to 30%. Therefore, efficiency of the gas turbine becomes less dependent on the compressor characteristics and the work ratio is considerably increased.

A practical concern with steam injection is water consumption, as total loss systems appear to be the normal arrangement. Consumption is, typically, between 1.1 and 1.6 kg of high purity water per kW h of electrical output. The necessary water purification system for large-scale plant would represent about 5% of the total capital expenditure and running costs would add about 5% to the fuel cost [10].

Because, in modern designs, the temperature of injected steam must be raised in the gas turbine combustion chamber to about 1250 °C, there are advantages in a close approach temperature in the waste heat boiler. However, because boiling is a constant temperature

process, high steam temperatures are necessarily associated with a low level of heat recovery. Conversely, the steam temperature must be relatively low if the heat recovery from the exhaust stream is to be maximised. This difficulty limits the maximum theoretical efficiency of the overall system [30]. In practice, the high temperature option is normally the best compromise [9].

4.2. The steam injected cycle with topping steam turbine

A steam injected gas turbine cycle with a topping [35] is shown in Fig. 17. A high-pressure steam is first expanded in a back-pressure stem turbine, producing power, and then is injected into the combustion chamber of the gas turbine.

4.3. The turbocharged steam injected cycle

The turbocharged steam injected gas turbine [6] is illustrated in Fig. 18. Such configuration would result in an average increase in power of 95% and an average efficiency rise from 30 to 42.6%.

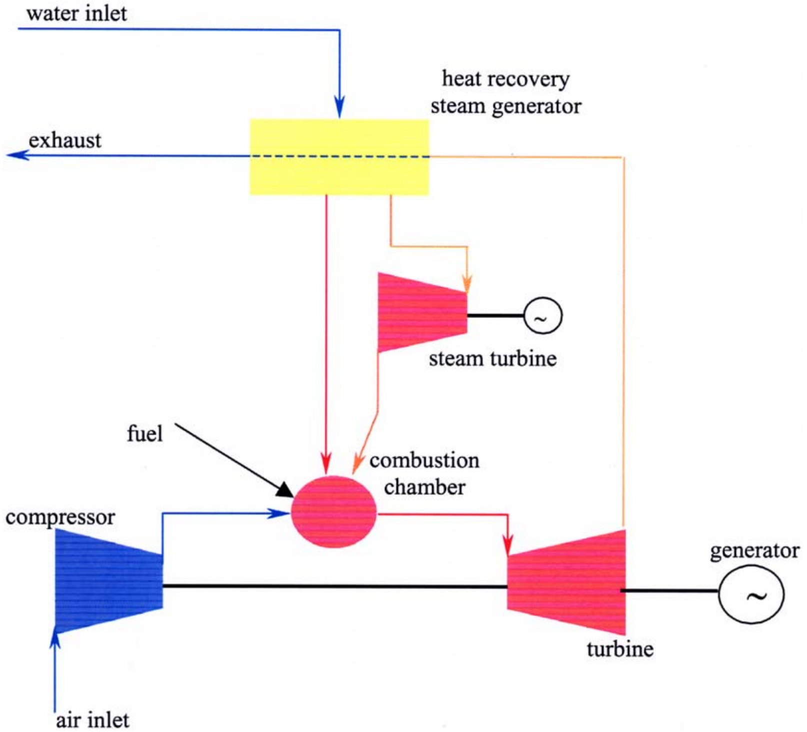

Fig. 17. The steam injected cycle with topping steam turbine.

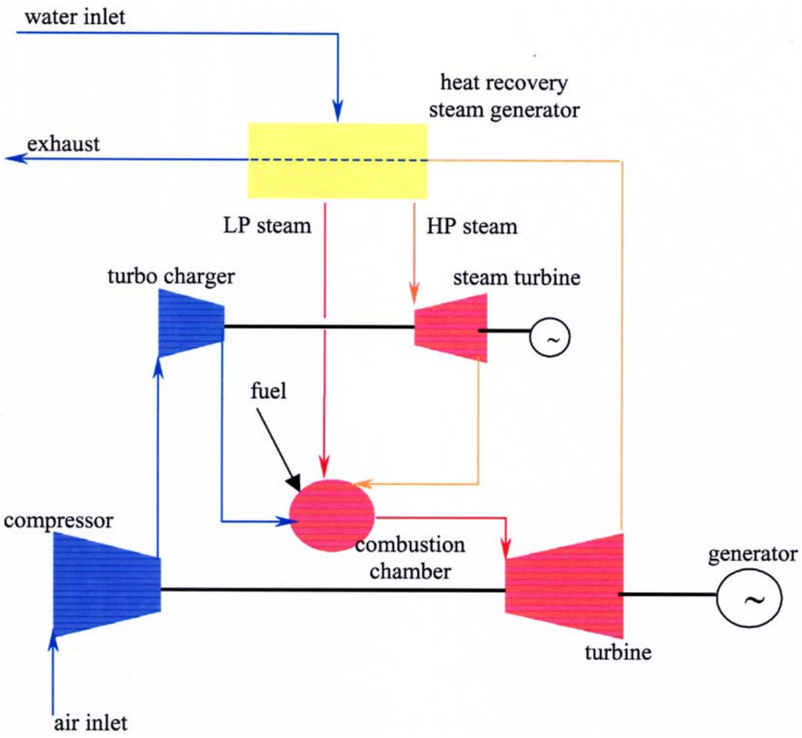

Fig. 18. The turbo charged steam injected cycle.

4.4. The DRIASI cycle

The dual-recuperated inter-cooled–after-cooled steam injected (DRIASI) cycle [3] combines steam injection, recuperation, and water injection as shown in Fig. 19. The analysis of this concept showed that the DRIASI cycle can provide comparable or superior efficiencies to those of conventional combined cycles for small systems up to 30 MWe. For large system, the performance of the DRIASI cycle was found to be inferior both to combined cycles and to steam injected turbines.

4.5. The evaporation cycle

The limitations of the waste heat boiler can be reduced with multi-pressure systems so that the successive saturation temperatures are matched more closely to those of the exhaust gases with a better approximation to a reversible process. However, steam injection systems cannot accommodate steam flows at different pressure without substantial design convolutions. Evaporation cycles overcome the boiler limitation problem by injecting liquid water into the gas turbine air flow at the compressor exit, as illustrated in Fig. 20. In effect, the heat of compression is used to evaporate the water and the resulting, single phase mixture is then heated by the turbine's exhaust gas in a suitable

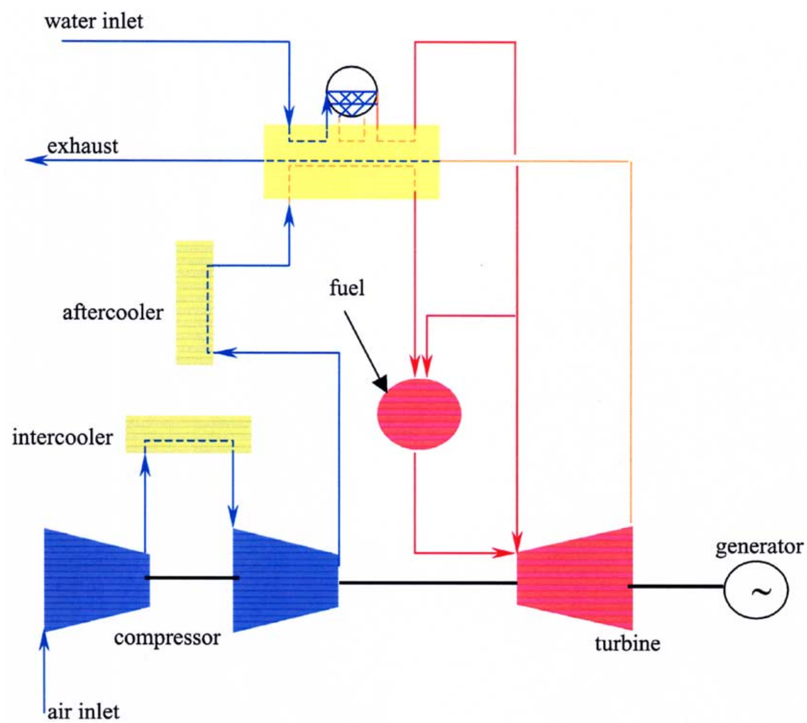

Fig. 19. The DRIASI cycle.

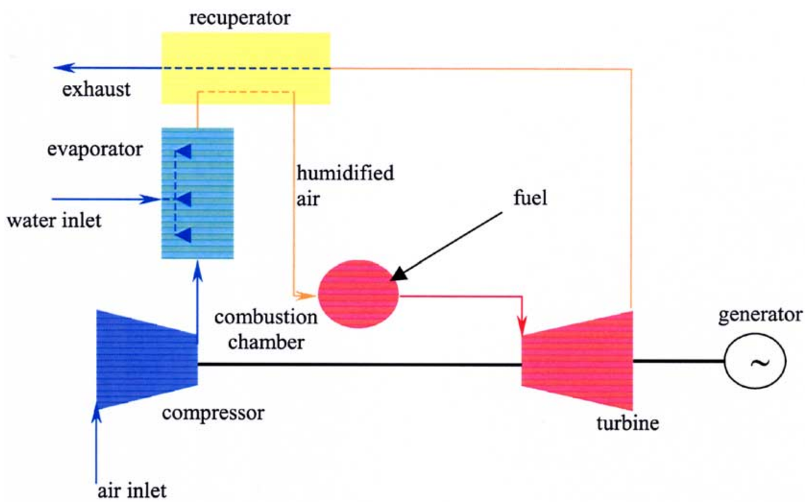

Fig. 20. The evaporation cycle.

heat exchanger. The benefits are the same as for steam injection, i.e. higher mass flow through the turbine and increased working fluid specific heat.

4.6. The HAT cycle

The best known of the advance evaporation cycles [33] is referred to as the humid air turbine (HAT) cycle. Originally proposed as the evaporative–regenerative cycle, the HAT cycle provides a substantial power boost of the system and an efficiency rise of several percentage points. A more advanced concept with inter-cooling, illustrated in Fig. 21, can provide even higher efficiency. The air is first compressed in the low-pressure compressor of the gas turbine and then enters the intercooler. The heat of compression is recovered for air saturation by circulating water and make-up water, which is passed to the saturator. Two main saturator types can be used in the HAT cycle, such as, plate towers and packing towers. In the former, the contact between the air and water flows is carried out by subsequent steps, because the liquid falls from one plate to the next. The latter exploits internal packing in order to enhance heat and mass exchange surface between the gas and water. The packing towers are characterised by lower pressure drops and lower capital cost, which will benefit HAT cycle applications. The saturator can operate with any clean and filtered water source as long as the dissolved substances at the water outlet remain below their precipitation concentration at the operating conditions. The water quality is maintained via a combination of the water treatment system and the saturator blow down to purge impurity [37].

The cooled compressed air is further compressed in the high-pressure compressor, cooled in the after-cooler and then fed to the air saturator. The air is contacted over

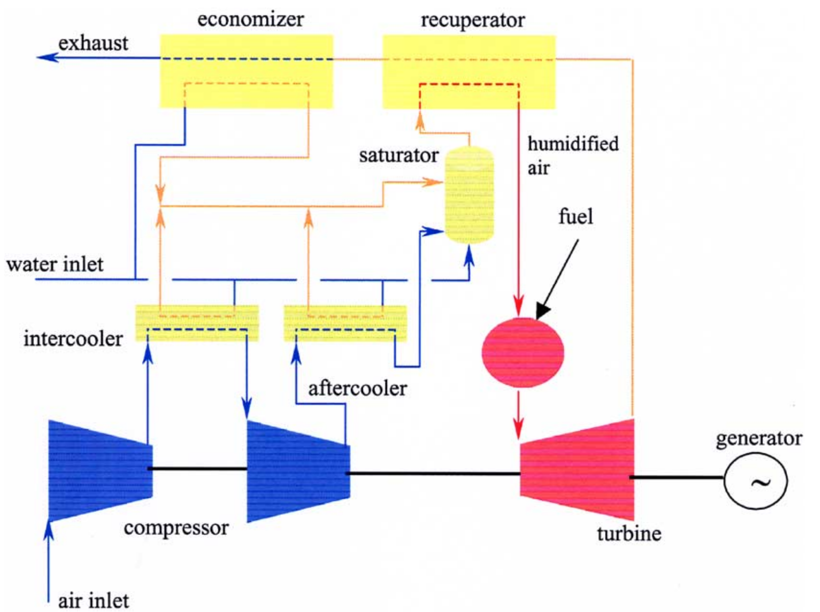

Fig. 21. The HAT cycle.

packing with water heated by the various heat sources. The humid air leaving the saturator, after preheating in the gas turbine exhaust, is fed to the combustion chamber. The hot gas exiting the combustion chamber expands through the turbine driving the compressors and providing power. The exhaust heat is then recovered in the recuperator and in the economizer to preheat the water for air saturation [33]. The water content increases the mass and the specific heat of the flow, which leads to additional power, and the use of the recuperator gives higher efficiency. By varying the water content the HAT cycle can be put in part-load operation without penalizing efficiency, and can also be started up in much shorter time than a conventional combined cycle plant. Water consumption is a problem for the same reasons as apply to steam injection cycles but the consumption rate is only about one-third.

Based on various studies, the efficiency of the HAT cycle varies from 54% for a low-pressure ratio turbine to 57% for a high-pressure cycle [37]. Further, the cycle does not require expensive steam/water equipment that simplifies the scheme and lowers operating and maintenance costs. A HAT cycle pilot project (600 kW) is currently under progress at the Lund University [23]. This is part of the evaporative gas turbine project, which involves Swedish Universities and main European gas turbine companies. The main aim of the project is the installation and experimentation of the first working HAT cycle in the world.

4.7. The LOTHECO cycle

Unlike other MAST cycles, where the bottoming steam cycle is eliminated and integrated into the gas turbine, the *low temperature heat* combined *cycle* (LOTHECO cycle) [5], shown in Fig. 22, uses an external energy source for the water-in-air

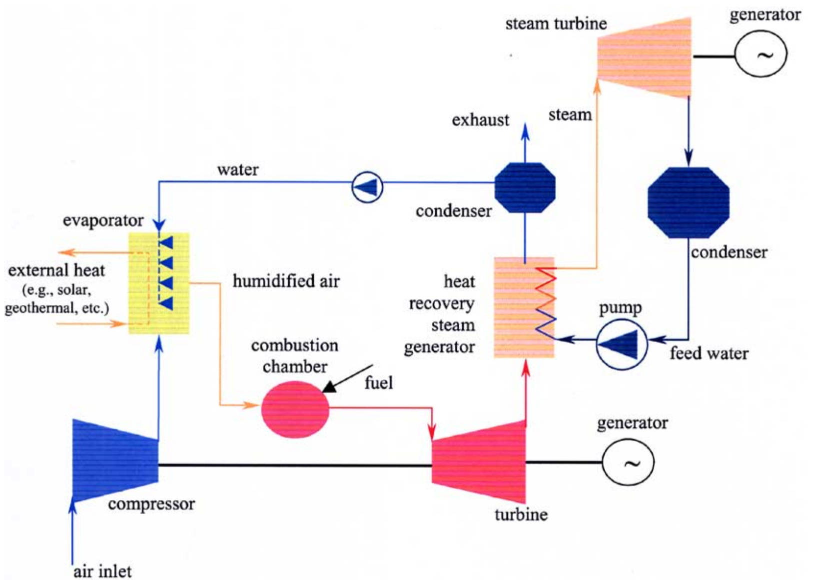

Fig. 22. The LOTHECO cycle.

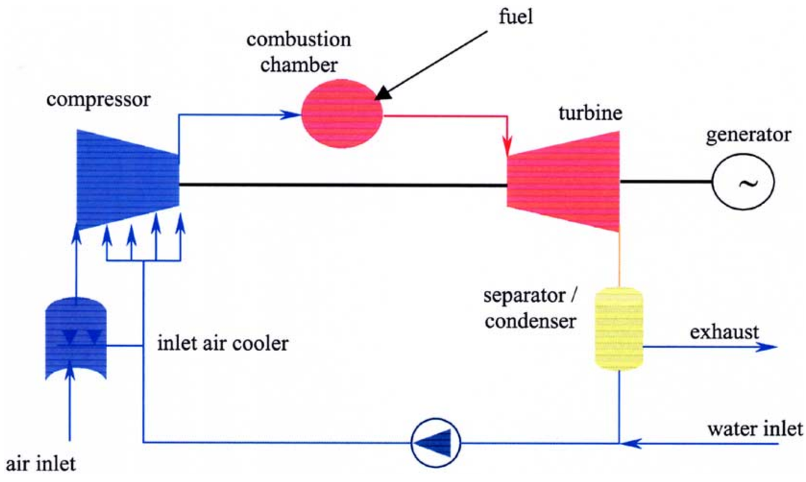

Fig. 23. The wet compression cycle.

evaporation, while the heat contained in the exhaust gas is utilised in a bottoming Rankine cycle. Since the water-in-air evaporation takes place at the vapour partial pressure, the saturation temperature is accordingly significantly low (below 170 °C). This temperature range (from below 100 °C up to 170 °C, depending on the amount of injected water and the compressor pressure ratio) is in favour of the integration of low-quality heat sources, that under other circumstances cannot be utilised for electric power generation such as, geothermal, solar, etc. These arrangements result in an enhanced fuel-to-electricity efficiency compared to the efficiency of an equivalent conventional combined cycle. Efficiencies above 60% have been recently reported [5].

4.8. The wet compression cycle

In the wet compression cycle, water injection is accomplished at the compressor stages, which results in nearly isothermal compression (Fig. 23). Water in the exhaust is recovered in a separating condensing unit. Such configuration can reach efficiency of 43% [28].

5. Discussion

Combined cycle technology, achieving efficiencies well over 58%, with plant capacities in the range between 350 and 500 MWe [31], is likely to remain, for many years, attractive as a new plant for utility scale applications [32]. The Kalina cycle can be regarded as a possible competitor to the combined cycle, however, it is largely untried and judgement must be reserved until there is some commercial operating experience.

The situation becomes more complex for relatively low capacity systems, say, less than about 50 MWe. Here, the choice is greater and different selection criteria may operate. For example, low capital cost may have greater emphasis than high efficiency and mechanical

complexity may be less acceptable and this type may favour steam injection for low capacity systems despite the economic advantage remaining with combined cycles. Furthermore, the potential performance of MAST technologies below 50 MWe capacity has been enhanced considerably in recent years by significant developments in the aero-derivative forms of industrial gas turbine [38]. A further important feature of MAST technologies is their compatibility with cogeneration which is the mode used in most commercial applications. For example, the Cheng cycle allows system optimisation under different heat and power demands.

Gas to gas recuperation does offer reasonable efficiency without mechanical complexity at relatively low capital cost. Furthermore, there is no requirement for large quantities of water for cooling or injection purposes. Although this system has a much lower absolute performance than the combined cycle technology, the variations in selection criteria, which tend to apply to smaller scale system, create niche market and the use of gas to gas recuperation may be a good solution to very specific requirements. The use of the fuel cells integrated with gas turbines allows efficiency to approach 70% and can, therefore, be seen as a choice for the future power plants [23].

6. Conclusions

In this work, an overview of the current and future sustainable MAST and non-MAST gas turbine technologies was carried out. In particular, the various MAST and non-MAST gas turbine technologies are described and compared. Emphasis has been given to the various advance cycles involving heat recovery from the gas turbine exhaust, such as, the gas to gas recuperation cycle, the combined cycle, the chemical recuperation cycle, the Cheng cycle, the humid air turbine cycle, etc. The thermodynamic characteristics of the various cycles were considered in order to establish their relative importance to future power generation markets.

Gas turbine technologies will play a major role in future power generation and several well-justified concepts have been developed or are the subject of major feasibility studies. The combined cycle technology is now well established and offers superior performance to any of the competing systems, which are likely to be available in the medium term for large-wale power generation applications. The Kalina cycle is a possible exception since there is some evidence to suggest that this scheme can provide higher efficiency without significant cost penalties. The combined cycle may be too complex mechanically for small-scale operations but there are a variety of schemes to help meet specific application requirement in this area.

In small-scale power generation, less than 50 MWe, it is more cost effective to install a less complex power plant, due to the adverse effect of the economics of scale. Combined cycle plants in this power output range have usually higher specific investment costs and lower electrical efficiencies but, on the other hand, robust and reliable performance. MAST technologies are among the possible ways to improve the performance of gas turbine based power plants at feasible costs (e.g. peak load gas turbine plants).

A promising future sustainable gas turbine technology is the Brayton–fuel cell cycle. A fuel cell system can operate at high pressure and can produce very high temperature

exhaust gases, which allows integrating a gas turbine within the system, thus improving performance. The Brayton–fuel cell cycle is claimed to have the highest efficiency of any advance cycle (approximately 70%), and can, therefore, be seen as a choice for the future power plants.

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